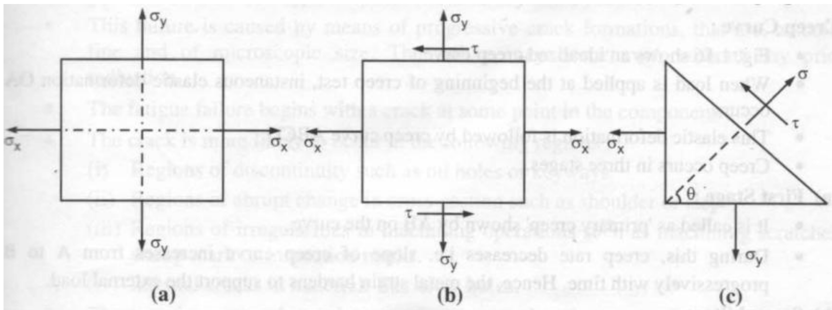
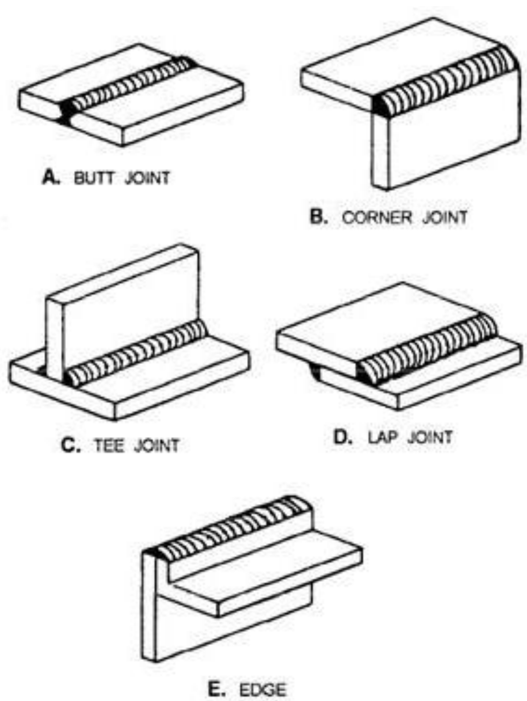
WINTER – 15 EXAMINATIONS

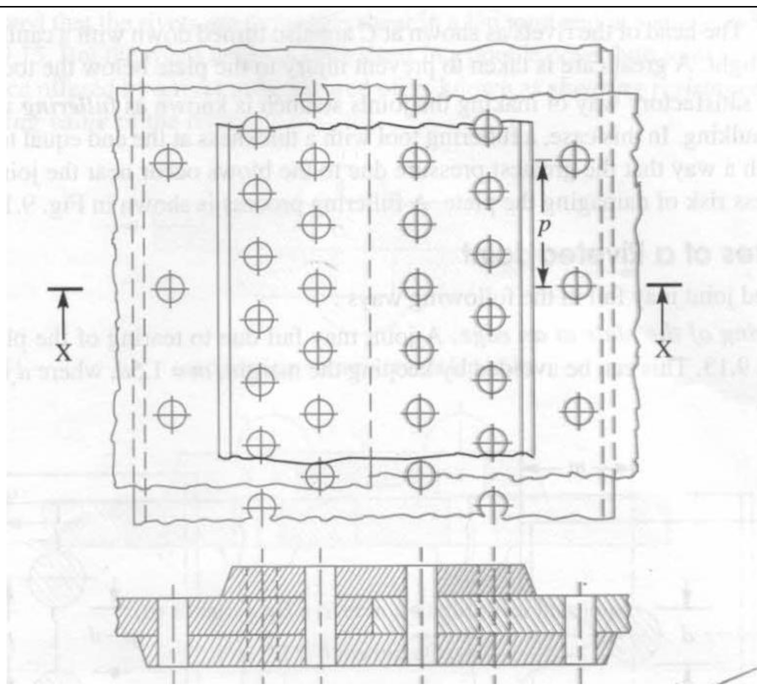
Subject Code: **17553** **Model Answer** Page No: ____/ N

Important Instructions to examiners:

1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling errors should not be given more importance. (Not applicable for subject English and Communication Skills)
4) While assessing figures, examiner may give credit for principal components indicated in the figure. The figures drawn by candidate and model answer may vary. The examiner may give credit for any equivalent figure drawn.
5) Credits may be given step wise for numerical problems. In some cases, the assumed constant values may vary and there may be some difference in the candidate's answers and model answer.
6) In case of some questions credit may be given by judgment on part of examiner of relevant answer based on candidate's understanding.
7) For programming language papers, credit may be given to any other program based on equivalent concept.

Q. NO.	MODEL ANSWER	MARKS	TOTAL MARKS
1	Attempt any FIVE of the following:		5X4=20
(a)	Maximum Normal Stress theory. 	01 mark for figure	
	•When the component is subjected to several types of loads simultaneously, it i necessary to determine the state of stresses under such conditions. For example, a transmission shaft is subjected to bending moment as well as twisting moment (torque) at the same time. • The plane, on which, only normal stresses acts and no shear stress, is called as principal plane. The magnitude of normal stress acting on the principal plane is called principal stress. • Consider an element of a plate subjected to two dimensional stresses as shown in Fig. • In this analysis, the stresses are classified into two groups : (a) Normal stress,(b) Shear stress. Normal stress is perpendicular to area under consideration, while shear stress acts over the area. Refer Fig. (c), showing the stresses acting on oblique plane.	01 mark	
	• "Major principal stress is the maximum value of normal stress acting on the principal plane, whereas, the minimum value of normal stress acting on principal plane is called asminor principal stress".This is called as **Maximum Normal Stress theory**or Principal Stress Theory**.**	01 mark	
	According to this maximum principal stresses are given as follows. $\sigma_{t1} = \{(\sigma_x + \sigma_y)/2\} + \frac{1}{2}\sqrt{\{[\sigma_x - \sigma_y]^2 + 4\tau^2\}}$ Minimum Principle stress, $\sigma_{t2} = \{(\sigma_x + \sigma_y)/2\} - \frac{1}{2}\sqrt{\{[\sigma_x - \sigma_y]^2 + 4\tau^2\}}$		04 marks
	Also maximum shear stress, $\tau_{max} = \{(\sigma_x - \sigma_y)/2\} = \frac{1}{2}\sqrt{\{[\sigma_x - \sigma_y]^2 + 4\tau^2\}}$	01 mark	
b	•Keyway is a slot machined either on the shaft or in the hub to accommodate the key. • It is cut by vertical or horizontal milling cutter. • The keyway cut into the shaft reduces the load carrying capacity of shaft.	01 mark	
	• This is due to stress concentration near the comers of the keyway and reduction in the crosssectionalarea of shaft. • In other words, the torsional strength of shaft is reduced.	01 mark	
	• The following relation of reduction factor is used to analyze the weakening effect of keyway is given by H. F. Moore.		

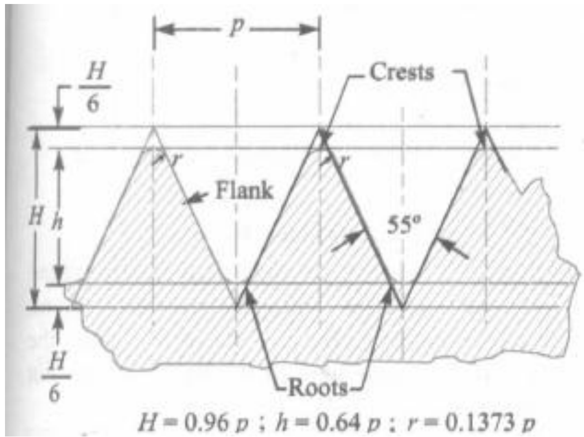
	e = 1 - 0.2 (w/d) - 1.1(h/d) Where, e = shaft strength factor = Strength of shaft with keyway/Strength Of shaft WIithout keyway w = Width of keyway, d = Diameter of shaft h = Depth of keyway = 112 x thickness of key = 1/2 x t • It is usually assumed that strength of keyed shaft is 75% of solid shaft. • Thus, after finding out dimensions of key, the reduction factor 'e' is calculated and for safe design, its value should be less than 0.75.	01 mark 01 mark	04 marks
c	The sketches of basic welding joints are given as follows.  The applications of the above joints are as follows. **Butt Joint** A butt weld, or a square-groove, is the most common and easiest to use. Consisting of two flat pieces that are parallel to one another, it also is an economical option. It is the universally used method of joining a pipe to itself, as well as flanges, valves, fittings, or other equipment. However, it is limited by any thickness exceeding 3/16”. **Corner Joint** A corner weld is a type of joint that is used between two metal parts and is located at right angles to one another in the form of a L. As the name indicates, it is used to connect two pieces together, forming a corner. This weld is most often used in the sheet metal industry and is performed on the outside edge of the piece. **Edge Joint**	02 marks for any four sketches	

	Edge welding joints, a groove type of weld, are placed side by side and welded on the same edge. They are the most commonly used type of joints due to build up accumulating on the edges. They are often applied to parts of sheet metal that have edges flanging up or formed at a place where a weld must be made to join two adjacent pieces together. **Lap Joint** This is used when two pieces are placed atop each other while also over lapping each other for a certain distance along the edge. Considered a fillet type of a welding joint, the weld can be made on one or both sides, depending upon the welding symbol or drawing requirements. It is most often used to join two pieces together with differing levels of thickness. **Tee Joint** Tee joints, considered a fillet type of weld, is used when two members intersect at 90° resulting in the edges coming together in the middle of a component or plate. It may also be formed when a tube or pipe is placed on a baseplate.	02 marks for any four applications.	04 marks
d	 Important Terms Used in Riveted Joints The following terms in connection with the riveted joints are important from the subject of view : (i)Pitch. It is the distance from the centre of one rivet to the centre of the next rivet measured parallel to the seam as shown in Fig. It is usually denoted by p. (ii)Back pitch. It is the perpendicular distance between the centre lines of	02 marks for figure	

	the successive rows as shown in Fig. It is usually denoted by Pb. (iii)Diagonal pitch. It is the distance between the centres of the rivets in adjacent rows of zig-zagriveted joint as shown in Fig. It is usually denoted by Pd. (iv)Margin or marginal pitch. It is the distance between the centre of rivet hole to the nearest edge of the plate as shown in Fig. It is usually denoted by m.	02 marks for 04 terms	04 marks

e

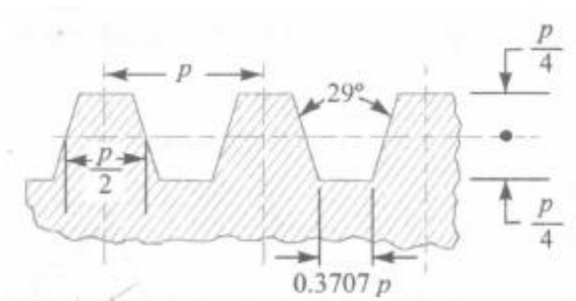
i) British standard whitworth (B.S.W) thread:



This is a British standard thread profile and has coarse pitches. It is a symmetrical V-thread in which the angle between the flankes, measured in an axial plane, is 55^0These threads are found on bolts and screwed fastenings for special purposes.The various proportions of B.S.W. threads are shown in Fig.

01 mark

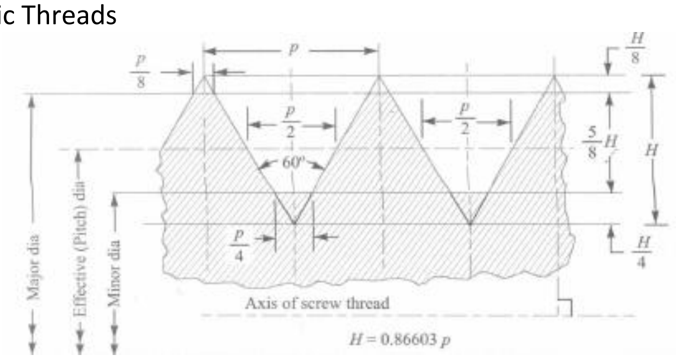
ii)Acme thread.

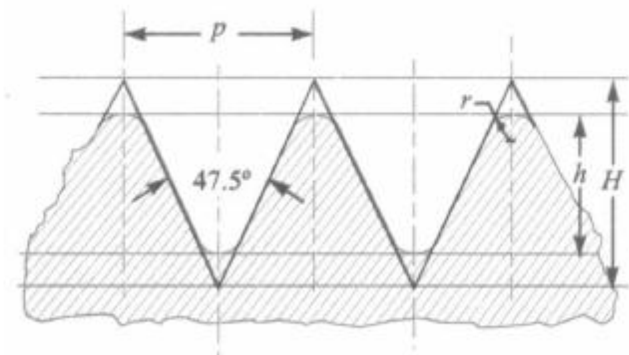


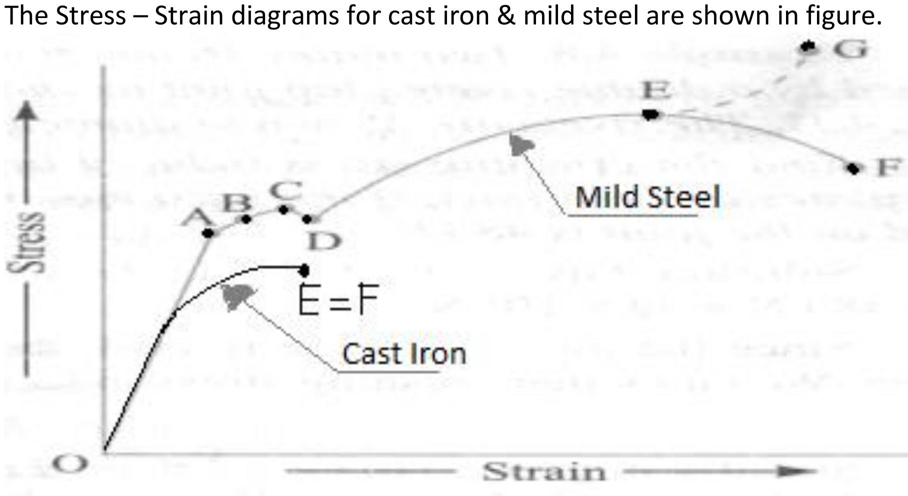
It is a modification of square thread. It is much stronger than square thread and can be easily produced. These threads are frequently used on screw cutting lathes, brass valves,cocks and bench vices.

01 mark

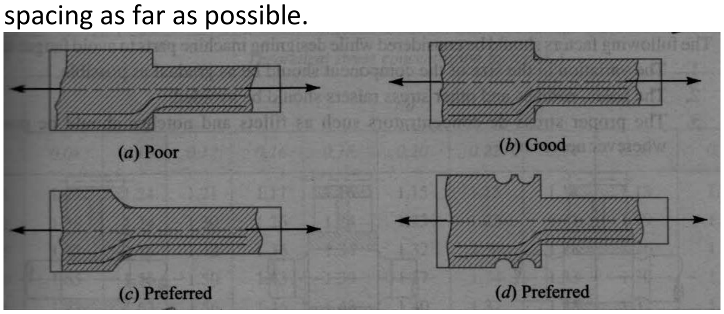
iii) Metric Threads



	It is an Indian standard thread and is similar to B.S.W. threads. It has an included angle of 60° instead of 55°. The basic profile of the thread is shown in Fig.a and the design profIle of the nut and bolt is shown in Fig. iv) British association (B.A.) thread.  $H = 1.13634\,p$; $h = 0.6\,p$; $r = 0.18083\,p$ This is a B.S.W. thread with fine pitches. The proportions of the B.A. thread are shown in Fig. These threads are used for instruments and other precision works.	01 mark 01 mark	04 marks

f	Stresses in Pipes The stresses in pipes due to the internal fluid pressure are determined by Lame's equation. According to Lame's equation, i) tangential stress at any radius x, $\sigma_t = \frac{p\,(r_i)^2}{(r_o)^2 - (r_i)^2}\left[1 + \frac{(r_o)^2}{x^2}\right]$ ...(i) and radial stress at any radius x, $\sigma_r = \frac{p\,(r_i)^2}{(r_o)^2 - (r_i)^2}\left[1 - \frac{(r_o)^2}{x^2}\right]$ ...(ii) where p = Internal fluid pressure in the pipe, ri = Inner radius of the pipe, and ro = Outer radius of the pipe. The tangential stress is maximum at the inner surface (when x = ri) of the pipe and minimum at the outer surface (when x = r) of the pipe. Substituting the values of x = r; and x = ro in equation (i), we find that the maximum tangential stress at the inner surface of the pipe, $\sigma_{t(max)} = \frac{p\,[(r_o)^2 + (r_i)^2]}{(r_o)^2 - (r_i)^2}$ and minimum tangential stress at the outer surface of the pipe, $\sigma_{t(min)} = \frac{2\,p\,(r_i)^2}{(r_o)^2 - (r_i)^2}$ The radial stress is maximum at the inner surface of the pipe and zero at the outer surface of the pipe. Substituting the values of x = ri and x = ro in equation (ii), we find that maximum radial stressat the inner surface, σr(max) =- p (compressive) and minimum radial stress at the outer surface of the pipe, σr(min) = 0	01 mark 01 mark 01 mark 01 mark	04 marks
g	The Stress – Strain diagrams for cast iron & mild steel are shown in figure. 	02 marks for figure	

	i) For Mild steel : A. Proportional limit:Hooke's law holds good up to point A and it is known as proportional limit. It is defined as that stress at which the stress-strain curve begins to deviate from the straight B. Elastic limit: The material has elastic properties up to the point B. This point is known as elastic limit. It is defined as the stress developed in the material without any permanent set C & D. Yeild Point: There are two yield points C and D. The points C and D are called the upper and lower yield points respectively. E. Ultimate stress: At E, the stress, which attains its maximum value is known as ultimate stress. ii) For Cast iron : F. Breaking strength: Failure without any indication.	01 mark 01 mark	04 marks
h	Following are the advantages and disadvantages of welded joints over other method joints. Advantages 1. The welded structures are usually lighter than riveted structures. This is due to the reason that in welding, gussets or other connecting components are not used. 2. The welded joints provide maximum efficiency (may be 100%) which is not possible in case of riveted joints. 3. Alterations and additions can be easily made in the existing structures 4. As the welded structure is smooth in appearance, therefore it looks pleasing. 5. In welded connections, the tension members are not weakened as in the case of riveted joints. 6. A welded joint has a great strength. Often a welded joint has the strength of the parent metal itself. 7. Sometimes, the members are of such a shape (i.e. circular steel pipe) that they afford difficulty for riveting. But they can be easily welded. 8. The welding provides very rigid joints. This is in line with the modern trend of providing rigid frames. 9. It is possible to weld any part of a structure at any point. But riveting requires enough clearance. 10. The process of welding takes less time than the riveting. Disadvantages 1.Since there is an uneven heating and cooling during fabrication, therefore the member may get distorted or additional stresses may develop. 2. It requires a highly skilled labour and supervision. 3. Since no provision is kept for expansion and contraction in the frame, therefore there is a possibility of cracks developing in it. 4. The inspection of welding work is more difficult than riveting work.	02 marks for any four advantages 02 marks for any two disadvantages	04 marks

2.	Attempt any TWO of the following:		2X8=16
(a) i	It is defined, in general, as the ratio of the maximum stress to the working stress. Mathematically, Factor of safety = Maximum stress/Working or design stress In case of ductile material e.g. mild steel, where the yield point is clearly defined, the factor of safety in based upon the yield point stress. In this case, Factor of safety =Yield point stress/Working or design stress In case of brittle material e.g. cast iron, the yield point is not well defined as for ductile materials. Therefore, the factor of safety for brittle materials is based on ultimate stress Factor of safety=Ultimate stress/ Working or design stress This relation may be used for ductile materials. The following things are considered for the selection of Factor of Safety. i) The type of product. (i.e. whether it is a utility good or machine part etc.) ii) The importance/ position of the component in the assembly. iii) The extent of damage to the people and/or to other parts that may take place due to the failure of the part. iv) The cost of the material.	01 mark 01 mark 02 marks	04 marks
ii	Stress concentration can be defined as the increase in the iintensity of stress due to various factors such as abrupt change in cross section, sharp corners,presence of holes, internal deformities, cracks,etc. The presence of stress concentration cannot be totally eliminated but it may be reduced to some extent. A device or concept that is useful in assisting a design engineer to visualize the presence of stress concentration and how it may be reduced is that of stress flow lines, as shown in Fig. The reduction of stress concentration means that the stress flow lines shall maintain their spacing as far as possible.  (a) Poor (b) Good (c) Preferred (d) Preferred	01 mark	

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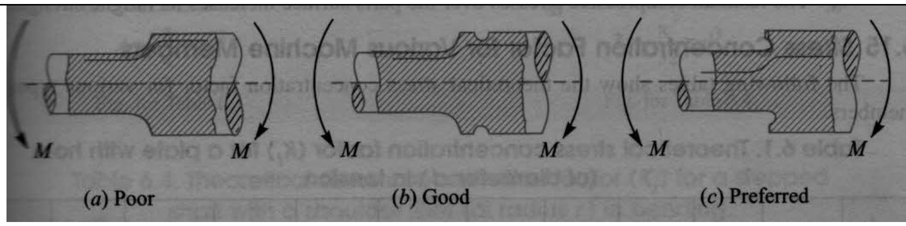
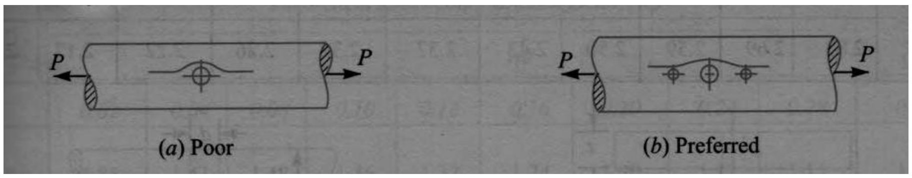
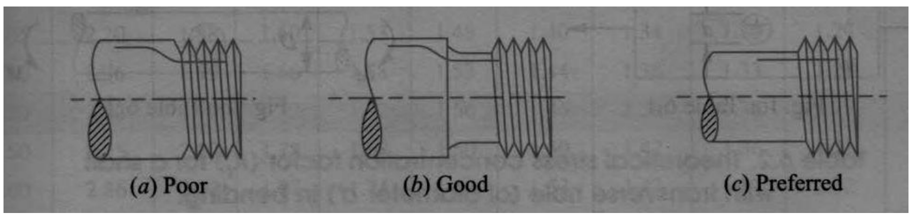
<tr><td></td><td>
Method of reducing stress contraction in cylinder members with shoulders


Method of reducing stress contraction in cylinder members with holes


Method of reducing stress contraction in threaded members with holes

The stress concentration effects of a press fit may be reduced by making more gradual transition from the rigid to the more flexible shaft. The various ways of reducing stress concentration for such cases are shown in Fig. a,b,c</td><td>01 mark

01 mark

01 mark</td><td>04 marks</td></tr>
<tr><td>b</td><td>Given data: $P = 8$ kW $= 8 \times 10^3$ W
N = 750rpm
$\tau s = 35$ MPa = 35 Nzmm^2
$\tau ci = 15$ N/mm^2, $\sigma t = \sigma ck = 60$ N/mm^2
The power transmitted by steel shafts,
$P= 2\pi NT /60$
TherforeTorque = $T = P \times 60 /2\pi N = 8 \times 103 \times 60 / 2 \times \pi \times 750$
. . $T = 101.859$ N-m $= 101.859 \times 10^3$ N-mm
i) Design of shaft :
We know that, torque transmitted by shaft is given by
$T= \pi/16 \times \tau s \times d^3$
$101.859 \times 10^3= \pi/16 \times 35 \times d^3$
Diameter of shaft, d = 25.56== 30 mm (say)
(ii)Design of hub: Usual proportions are,
D = Outer diameter of hub
= 2d = 2 x 30 = 60 mm
L = Length of hub = 1.5 x d = 1.5 x 30 = 45 mm
k = d/D = 30/60 = 0.5
Considering hub as a hollow shaft transmitting the same torque as that of shaft.Then we have</td><td>01 mark

01 mark</td><td></td></tr>
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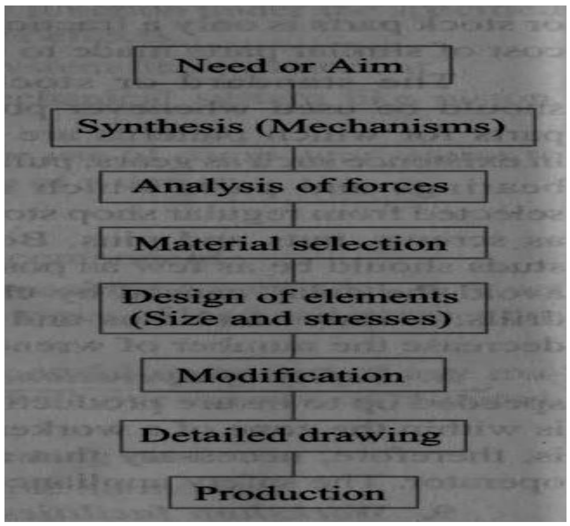
	$T = \pi/16 \times \tau_{ci} \times D^3(1-k^4)$ $101.859 \times 10^3 = \pi/16 \times \tau_{ci} \times 60^3(1-0.5^4)$ $\tau_{ci} = 2.561\ N/mm^2$ Thus, the induced shear stress in the cast iron hub is less than the given permissible shear stress. Hence, the design is safe. (iii) Design of flange: Take tf=d / 2 = 30 /2 = 15 mm While transmitting the torque, the flange is under shear. The torque transmitted is T = Circumference of hub x Thickness of flange x Shear stress x Radius of hub $=(\pi \times D) \times t_f \times \tau_f \times D/2$ $101.859 \times 10^3 = (\pi \times 60) \times 15 \times \tau_f \times 60/2$ $\tau_f = 1.2\ N/mm^2$ Thus, induced shear stress is less than given permissible shear stress for flange material.Hence, the design is safe. iv) Design of key: It is nothing but checking the safety of the key in shear & crushing. For the shaft of 30 mm dia recommended size of key is w=10mm & t=08 mm Checking the key in shear : We know, Torque transmitted by the shaft, $T = L \times w \times \tau \times (d/2)$ where τ is the induced stress in key material. Hence, $101.859 \times 10^3 = 45 \times 10 \times \tau \times 30/2$ $\tau = 15.09\ N/mm^2$ which is less than τ_s (35 MPa) Hence the key is safe in shear. Similarly checking the key for crushing, $T = L \times (t/2) \times \sigma_c \times (d/2)$ Hence, $101.859 \times 10^3 = 45 \times 8/2 \times \sigma_c \times 30/2$ $\sigma_c = 37.72$ Mpa which is less than $\sigma_{ck} = 60\ N/mm^2$ Hence the key is safe in crushing..	02 marks 02 marks 01 mark 01 mark	08 marks
(c)	i) The **general procedure**in machine design is as follows: 1. Recognition of need: First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed. 2. Synthesis (Mechanisms): Select the possible mechanism or group of mechanisms which will give the desired motion. 3. Analysis of forces: Find the forces acting on each member of the machine and the energy transmitted by each member. 4. Material selection: Select the material best suited for each member of the machine. 5. Design of elements (Size and Stresses):Find the size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. It should be kept in mind that each member should not deflect or deform than the permissible limit.	01 mark each for any four points.	

6. Modification: Modify the size of the member to agree with the past experience and judgement to facilitate manufacture. The modification may also be necessary by consideration of manufacturing to reduce overall cost.
7. Detailed drawing: Draw the detailed drawing of each component and the assembly of the machine with complete specification for the manufacturing processes suggested.
8. Production: The component, as per the drawing, is manufactured in the workshop.

General procedure in Machine Design.



ii) The **general considerations** in machine design are as follows.
01)Type of Load and Stresses caused by the Load:-
The load on the Machine Component,may act in several ways due to which the Internal Stresses are set up.

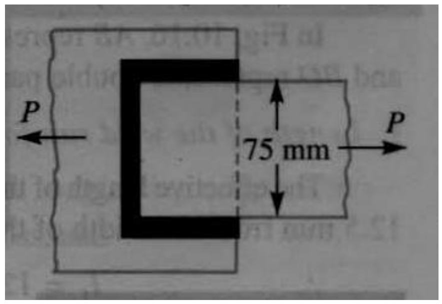
02)Motion of Parts:-
The successful operation of any Machine depends largely upon the simplest arrangements of the Parts,which will give the required motion.The Motion of the Part may be
A)RectilinearMotion,which includes Unidirectional and Reciprocating Motion.
B)CurvilinearMotion,which includes Rotary,Oscillatory Simple Hormonic.
C)Constant Velocity.
D)Constant or Variable Acceleration.

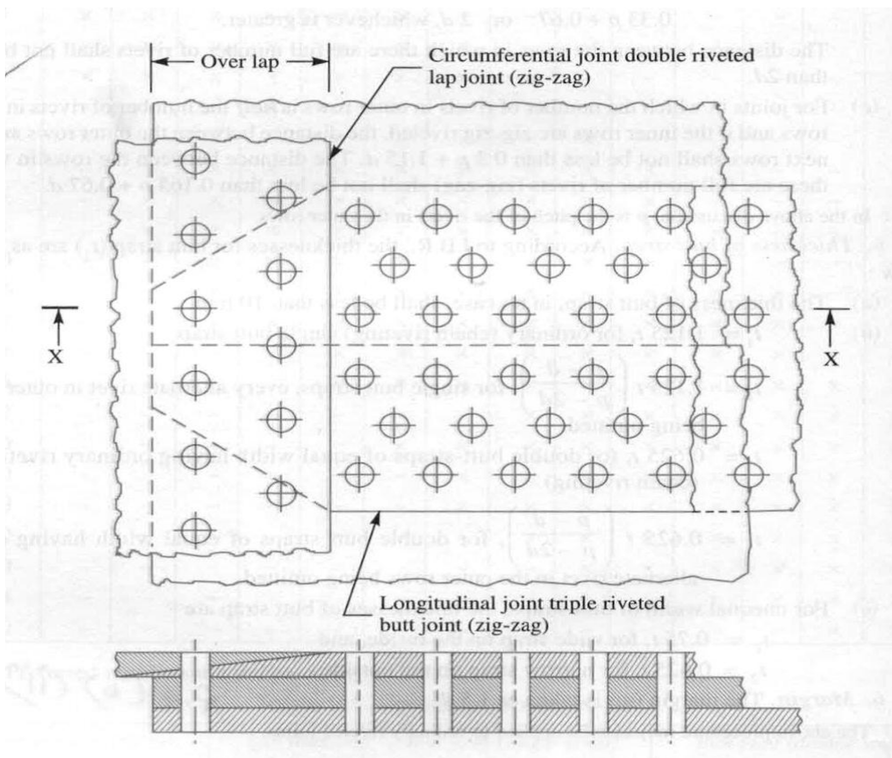
03)Selection of Material:-
Every Machine Design Engineer should have a thorough knowledge of the Properties of Material and their behaviour under working conditions.

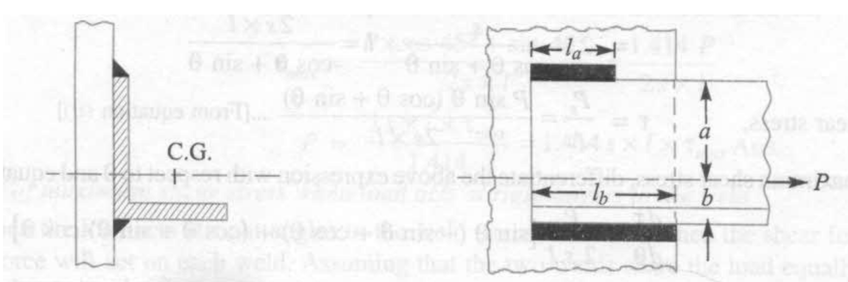
04)Form and Size of the Parts:-
In order to design any Machine Part for form and size,it is necessary to

	know the Forces which the Part must sustain.Any suddenly applied or impact load must be taken into consideration,which may cause failure.The smallest Practicable Cross-Section may be used,but it may be checked that the Stresses induced in the Designed Cross-Section are reasonably safe. 05)Frictional Resistance and Lubrication:- There is always a Loss of Power due to Frictional Resistance.Careful attention must be given to the matter of Lubrication of all surfaces which moves in contact with others. 06)Safety of Operator:- A Machine Designer should always provide safety device for the safety of the operator.The Safety Appliances should in no way interfere with the operation of the Machine. 07)Use of Standard Parts:- The use of Standard Parts are closely related to the Cost of Machine,because the Cost of Standard Parts is only a fraction of the cost of similar parts made to order. 08)Convenient and Economical Features:- The operating feature of the Machine should be carefully studied.TheStarting,Controlling and Stopping Levers should be located on the basis of convenient handling. 09)Workshop Facilities:- A Design Engineer should be familiar with limitation of his Employer's Workshop,in order to avoid the necessity of having work-done in some other Workshop. 10)Assembling:- Every Machine must be Assembled as a unit before it can function.The final Location of any Machine is important and the Design Engineer must anticipate the exact location and the local facilities for erection. Above considerations are most important in machine design engineering.	01 mark each for any four points.	08 marks
3.	Attempt any TWO of the following:		2X8=16
(a)	Given K=0.8;P=400KW;N=225RPM;M=5000N.m;τ=50MPa Solution T=60P / 2πN =(60 x 400 x 10^3)/(2 x π x 225) =16976.52N.M Te=Euivalanttwisting moment =√(M^2 + T^2) =√(5000^2 + 16976.52^2) =17697.52N.m	 01 mark 01 mark	

	$=17697.52 \times 10^3$ N.mm We know, $Te=(\pi/16) \times \tau(do)^3(1-k^4)$ $17697.52 \times 10^3=(\pi/16) \times 50(do)^3(1-0.8^4)$ $do^3=17697.52 \times 10^3 \times 16 / \pi \times 50$ $do^3=\sqrt[3]{(3053276.735)}$ do=145.07mm say 150mm (Since generally shafts in this range are manufactured in the slabs of 5 mm) Ans.	01 mark 01 mark 02 marks 01 mark 01 mark	08 marks
(b)	A key connecting the shaft and hub Let T = Torque transmitted by the shaft, F = Tangential force acting at the circumference of the shaft, d = Diameter of shaft, l = Length of key, w = Width of key. t = Thickness of key, and τ and σc = Shear and crushing stresses for the material of key. Due to the power transmitted by the shaft, the key may fail due to shearing or crushing. Considering shearing of the key, the tangential shearing force acting at the circumference of the shaft, F = Area resisting shearing x Shear stress = $l \times w \times \tau$ Torque transmitted by the shaft, $T= F\times (d/2) = l w \tau \times (d/2)$ ...(i) Considering crushing of the key, the tangential crushing force acting at the circumference of the shaft, F = Area resisting crushing x Crushing stress = $l \times (t/2) \times \sigma c$ Torque transmitted by the shaft, $T= F\times (d/2) = l \times (t/2) \times \sigma c \times (d/2)$ ...(ii) The key is equally strong in shearing and crushing, if $l w \tau \times (d/2) = l \times (t/2) \times \sigma c \times (d/2)$ ..[Equating equations (I) and (ii)] Or $w/t = \sigma c / 2\tau$ The permissible crushing stress-for the usual key material is atleast twice the permissible shearing stress. Therefore from equation (iii), we have w = t. In other words, a square key is equally strong in shearing and crushing.	01 mark 02 marks 02 marks 01 mark 02 marks	08 marks

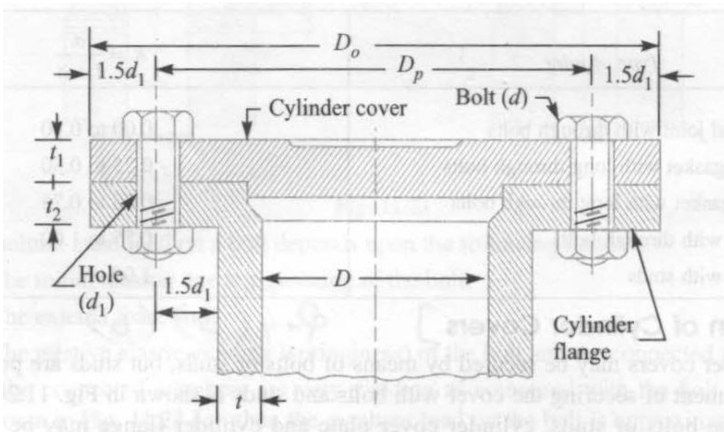
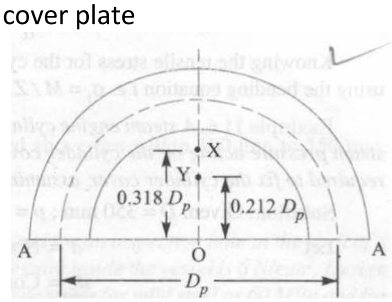
(c)	 Given: Width = 75 mm; Thickness = 12.5 mm; σ_t = 70 MPa = 70 N/mm^2 ; t = 56 MPa = 56 N/mm^2. The effective length of weld L1 for the transverse weld may be obtained by subtracting 12.5 mm from the width of the plate. L1 = 75 - 12.5 = 62.5 mm Length of each parallel fillet for static loading Let L2 = Length of each parallel fillet We know that the maximum load which the plate can carry is P = Area x Stress = 75 x 12.5 x 70 = 65 625 N Load carried by single transverse weld, P1= 0.707 s x L1 Xσ_t= 0.707 x 12.5 x 62.5 x 70 = 38 664 N and the load carried by double parallel fillet weld, P2= 1.414 s x L2 X T= 1.414 x 12.5 x 12 x 56 = 990L2 N Load carried by the joint (P), 65625 = P1 + P2 = 38 664 + 990 L2 so L2 = 27.2 mm Adding 12.5 mm for starting and stopping of weld run we have L2= 27.2 + 12.5 = 39.7 says 40 mm Ans.	01 mark for figure 01 mark 01 mark 01 mark 01 mark 01 mark 01 mark 01 mark	08 marks

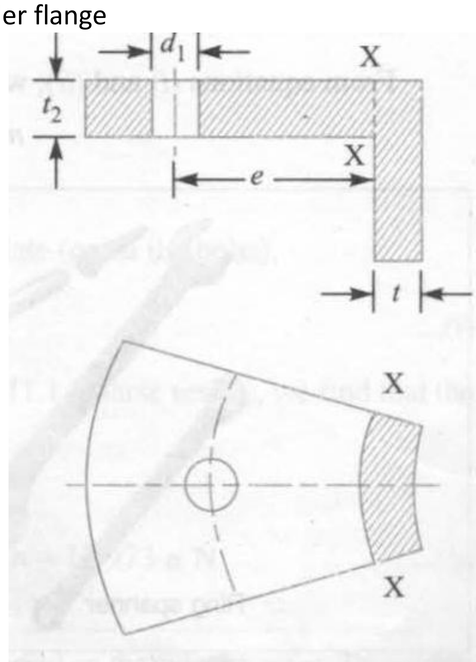
4.	Attempt any TWO of the following:		2X8=16
(a)	 Design of Circumferential Lap Joint for a Boiler The following procedure is adopted for the design of circumferential lap joint for a boiler. 1. Thickness of the shell , t =(P.D/2stxη)+1mm where P= pressure inside the boiler shell , D =dia. of the boiler shell , st = permissible tensile stress of the boiler plate η = efficiency of the joint. 2. diameter of rivet, d = 6√t 3. Number of rivets. Since it is a lap joint, therefore the rivets will be in single shear. Shearing resistance of the rivets,	 01 mark 01 mark	

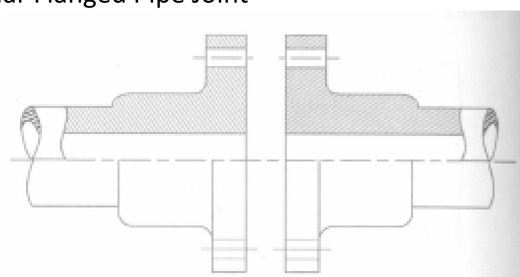
	$Ps = n \times \pi/4 \times d^2 \times \tau$... (i) wheren = Total number of rivets. Knowing the inner diameter of the boiler shell (D), and the pressure of steam (P), the total shearing load acting on the circumferential joint, $Ws = \pi/4 \times D^2 \times P$ -----(ii) From equations (i) and (ii), we get $n \times \pi/4 \times d^2 \times \tau = \pi/4 \times D^2 \times P$ $n = (D/d)^2 \times (P/\tau)$	01 mark	
	4. Pitch of rivets. If the efficiency of the longitudinal joint is known, then the efficiency of the circumferential joint may be obtained. It is generally taken as 50% of tearing efficiency in longitudinal joint, but if more than one circumferential joints is used, then it is 62% for the intermediate joints. Knowing the efficiency of the circumferential lap joint (TIc)' the pitch of the rivets for the lap joint (P1) may be obtained by using the relation: $\eta c = (P1 - d)/P1$	01 mark	
	5. Number of rows**.** The number of rows of rivets for the circumferential joint may be obtained from the following relation: Number of rows = Total number of rivets /Number of rivets in one the number of rivets in one row $\pi(D + t)/p1$ whereD = Inner diameter of shell.	01 mark	
	6. After finding out the number of rows, the type of the joint (i.e. single riveted or double riveted etc.) may be decided. Then the number of rivets in a row and pitch may be re-adjusted. In order to have a leak-proof joint, the pitch for the joint should be checked from Indian Boiler regulations.	01 mark 01 mark	
	7. margin , m = 1.5 d where d = dia. of rivet. 8. After knowing the distance between the rows of rivets (Pb)' the overlap of the plate may be fixed by using the relation, Overlap = (No. of rows of rivets - 1) Pb + m where m = Margin	01 mark	08 marks
(b)	 Letl_a= Length of weld at the top, l_b = Length of weld at the bottom, l= Total length of weld = la + lb P = Axial load, a = Distance of top weld from gravity axis,	02 marks for figure	

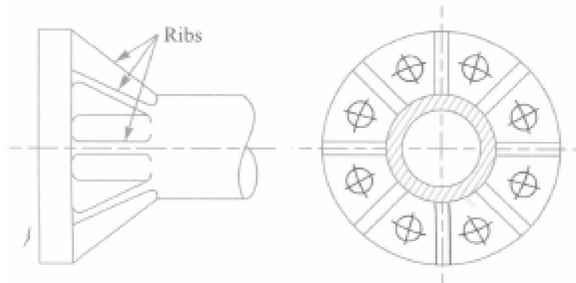
	b = Distance of bottom weld from gravity axis, and f = Resistance offered by the weld per unit length. Moment of the top weld about gravity axis = l_a x f x a and moment of the bottom weld about gravity axis . = l_b xfx b Since the sum of the moments of the weld about the gravity axis must be zero, therefore, l_ax f x a= l_b xfx b ……..(i) We know that l= l_a+ l_b …….(ii) From equations (i) and (ii), we have . l_a = (l x b)/(a + b) and . l_b = (l x b)/(a + b)	01 mark 01 mark 01 mark 01 mark 01 mark 01 mark	08 marks
(c)	Given: P = 2.5 N/mm^2; D = 1.6 m = 1600 mm .σt= 75 MPa = 75 N/mm^2: , τ = 60 MPa = 60 N/mm^2: , σc = 125 MPa = 125 N/mm^2 Design of circumferential joint The circumferential joint for a steam boiler may be designed as follows: 1. The thickness of the boiler shell (t) , is calculated as t =(P.D/2stxη)+1mm where P= pressure inside the boiler shell , D =dia. of the boiler shell , st = permissible tensile stress of the boiler plate η = efficiency of the joint. Hence, t = (2.5*1.6/2*75)+1mm = 27.6 say **28 mm**. 2. dia. of rivet d = 6√t =6√28 = 31.71 mm . Hence the rivet of 33mm dia is selected. 3.Number of rivets Let n = Number of rivets. We know that shearing resistance of the rivets =n x(π/4) xd^2 x τ …………………(i) and total shearing load acting on the circumferential joint =(π/4) xD^{2}x P ……………………….. (ii) From equations (i) and (ii), we get n x(π/4) xd^2 x τ =(π/4) x D^2 x P n=(D^2 x P)/(d^2 x τ)=(1600^2 x 2.5)/(34.5^2 x 60)= 89.6 say 90 Ans. 4. Pitch of rivets Assuming the joint to be double riveted lap joint with zig-zag riveting, therefore number of rivetsper row = 90/2 = 45 We know that the pitch of tile rivets, P1=[π(D+t)] /Number of rivets per row= π (1600+28) /45 = 113.7 mm	01 mark 01 mark 02 marks	

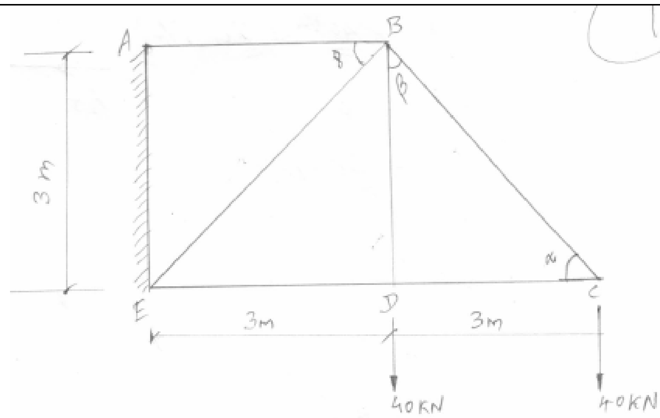
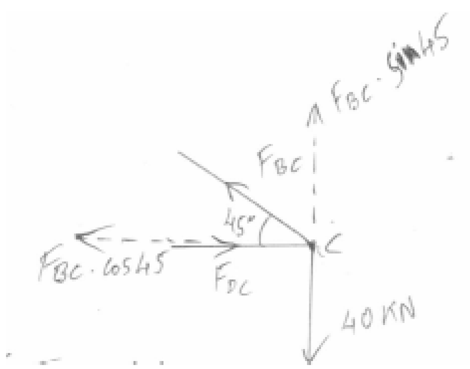
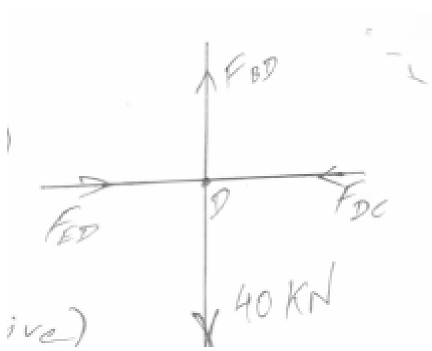
	Let us take pitch of the rivets, P_1 = 140 mm Ans. 5. Efficiency of the joint We know that the efficiency of the circumferential joint, $\eta_c = (P_1 - d) / p_1 = (140 - 34.5) / 140 = 0.753$ or 75.3% 6. Distance between the rows of rivets We know that the distance between the rows of rivets for zig-zag riveting, $= 0.33\ P_1 + 0.67\ d = 0.33 \times 140 + 0.67 \times 34.5$ mm = 69.3 say 70 mm Ans. 7. Margin We know that the margin, $m = 1.5\ d = 1.5 \times 34.5$ = 51.75 say 52 mm Ans.	01 mark 01 mark 01 mark 01 mark	 08 marks

5.	Attempt any TWO of the following:		2X8=16
(a)	Design of Cylinder Covers 1. Design of bolts or studs  (a) Arrangement of securing the cylinder cover with bolts. In order to find the size and number of bolts or studs, the following procedure may be adopted. Let D = Diameter of the cylinder, p = Pressure in the cylinder, dc = Core diameter of the bolts or studs, n = Number of bolts or studs, and бtb= Permissible tensile stress for the bolt or stud material. We know that upward force acting on the cylinder cover, $P = \pi/4\ (D)^2$ -------------(i) This force is resisted by n number of bolts or studs provided on the cover. Therefore Resisting force offered by tt number of bolts or studs $P = \pi/4\ (dc)^2$бtb X n -------------(ii) From equations i) and (ii), we have $\pi/4\ (D)^2 = \pi/4\ (dc)^2$бtb X n The tightness of the joint also depends upon the circumferential pitch of the bolts or tuds. The circumferential pitch should be between 20 √d1 and 30 √d2 ,where d) is the diameter of the hole in mm for bolt or stud. The pitch circle diameter (Dp) is usually taken as D + 2t + 3d1 and outside diameter of the cover is kept as Do = Dp + 3d1 = D + 2t + 6d1 where t = Thickness of the cylinder wall 2. Design of cylinder cover plate 	01mark for figure 01 mark 01 mark	

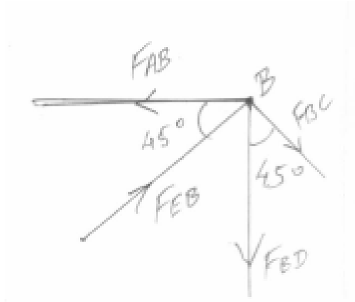
	Semi cover plate of cylinder The thickness of the cylinder cover plate (t)) and the thickness of the cylinder flange (t2) may be determined as discussed' below: We know that the bending moment at A-A, M = (Total bolt load/2)(OX- OY)= P/2(0.318Dp – 0.212Dp) = (P/2) X 0.106Dp = 0.053P X Dp	01 mark	
	Section modulus, $Z = (1/6)w(t1)^2$ where w = Width of plate = Outside dia. of cover plate - 2 x dia. of bolt hole =Do-2d1 Knowing the tensile stress for the cover plate material, the value of t1, may be determined by using the bending equation, i.e. бt =M/Z.	01 mark	
	3.Design of cylinder flange  A portion of cylinder flange. The thickness of the cylinder flange (t2) may be determined from bending consideration. A portion of the cylinder flange under the influence of one bolt is shown in Fig. The load in the bolt produces bending stress in the section X-X. From the geometry of the figure, we find that eccentricity of the load from section X-X is e = Pitch circle radius - (Radius of bolt hole +Thickness of cylinder wall) = (Dp / 2) – {(d1/2)+ t} Bending moment, M = Load on each bolt x e = P/nx e Radius of the section X-x, R = Cylinder radius + Thickness of cylinder wall = D/2 + t	01 mark for figure	

	Width of the section X-X, w = 2π R /n where n is the number of bolts. Section modulus, $Z = 1/6 \times w(t2)^2$ Knowing the tensile stress for the cylinder flange material, the value of t2 may be obtained by using the bending equation i.e. бt =M / Z	01 mark 01 mark	08 marks
(b)	Design of Circular Flanged Pipe Joint  Fig. 1 Cosider a circular flanged pipe joint as shown in Fig.1. In designing such joints, it is assumed that the fluid pressure acts in between the flanges and tends to separate them with a pressure existing at the point of leaking. The bolts are required to take up tensile stress in order to keep the flanges together. The effective diameter on which the fluid pressure acts, just at the point of leaking, is the diameter of a circle touching the bolt holes. Let this diameter be D 1. If d1 is the diameter of bolt holeandDp is the pitch circle diameter, then D1 = Dp-d1 :. Force trying to separate the two flanges, Pipes and PJpe Joints $F= \pi/4(D1)^2 \times P$ ……………………(i) Let n = Number of bolts, dc = Core diameter of the bolts, and σt = Permissible stress for the material of the bolts. .. Resistance to tearing of bolts $= \pi/4 \times (dc)^2 x\ n$ ...(ii) Assuming the value of dc the value of n may be obtained from equations (i) and (ii). The number of bolts should be even because of the symmetry of the section. The circumferential pitch of the bolts is given by P =(πDp)/n In order to make the joint leakproof, the value of Pc, should be between 20 √d1 to 30 √d1:, where d1 is the diameter of the bolt hole. Also a bolt of less than 16 mm diameter should never be used	01 mark 01 mark	

	to make the joint leakproof.  Fig. 2		
	The thickness of the flange is obtained by considering a segment of the flange as shown in Fig. 2. In this it is assumed that each of the bolt supportsone segment. The effect of joining of thesesegments on the stresses induced is neglected. The bending moment is taken about the section X-X,which is tangential to the outside of the pipe. Let the width of this segment is x and the distance of thissection from the centre of the bolt is y.	01 mark 01 mark	
	.. Bending moment on each bolt due to the force F =(F/n) x y ………………...(iii) and resisting moment on the flange	01 mark	
	= σb x Z …………..(iv) Whereσb = Bending or tensile stress for the flange material, and Z = Section modulus of the cross-section of the flange = 1/6 x $(t\ f)^2$ Equating equations (iii) and (iv), the value of If may be obtained. The dimensions of the flange may be fixed as follows: Nominal diameter of bolts, d = 0.75 t+ 10 mm Number of bolts, n = 0.0275 D + 1.6 ...(D is in mm) Thickness of flange, tf = 1.5 t + 3 mm Width of flange, B = 2.3 d	01 mark	
	Outside diameter of flange, Do = D + 2t + 2B Pitch circle diameter of bolts, Dp = D + 2d+ 2t+ 12 mm	01 mark 01 mark	08 marks

(c)	 Using joints Method, tan α = BD/DC = 3/3 Therforeα $=\tan^{-1}1$ $\alpha=45^0$ At jointC  ΣFy=0 $FBC.Sin45^0=40$ $FBC=40/Sin45^0$ FBC=56.56KN(Tensile) ΣFx=0 $FDC=FBC.cos45^0$ FDC=56.56 x $\cos45^0$=40KN(Compressive) At Joint D  ΣFy=0 FBD=40(Tensile)	01 mark 01 mark 01 mark 01 mark	

ΣFx=0
FED=40KN(Tensile)
At joint B, — 01 mark



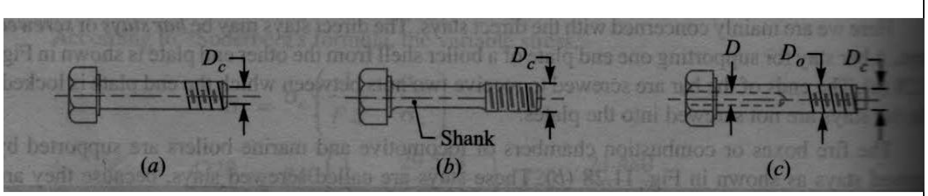
$\tan\beta = DC/BD = 3/3$
$\beta = \tan^{-1}1 = 45^0$
$\tan\gamma = AE/AB = 3/3$
$\gamma = \tan^{-1}1 = 45^0$
ΣFy=0
$FEB.\sin45^0 - 40 - FBC.\cos45^0 = 0$
$FEB.\sin45^0 - 40 - 56.56.\cos45^0 = 0$
$FEB.\sin45^0 = 80KN$
$FEB = 80/\sin45^0 = 11.12KN$(Compressive)
ΣFx=0 — 01 mark
$-FAB + FEB.\cos45^0 + FBC.\sin45^0 = 0$
$-FAB + 113.12.\cos45^0 + 56.56.\sin45^0 = 0$
FAB=119.98KN(Tensile)
Force table — 01 mark

Sr.NO.	Member	Force in KN	Nature
1.	AB	119.98	Tensile
2.	BC	56.56	Tensile
3.	CD	40	Compressive
4.	DE	40	Compressive
5.	BD	40	Tensile
6.	BE	113.12	Compressive

01 mark — 08 marks

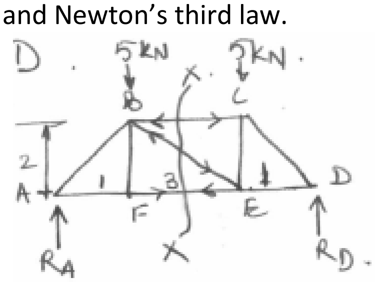
6. Attempt any **TWO** of the following: — 2X8=16

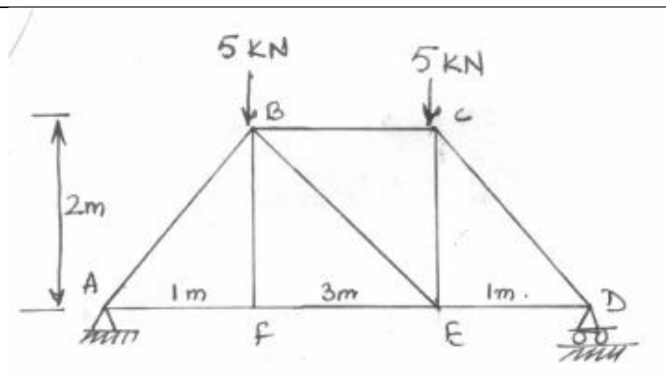
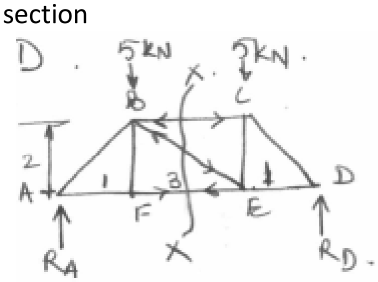
(a)i Bolt of uniform strength

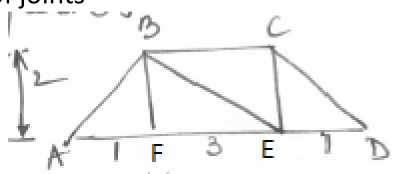
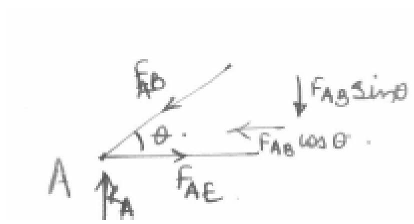
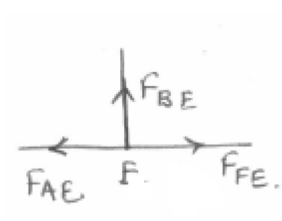


01 mark for figure

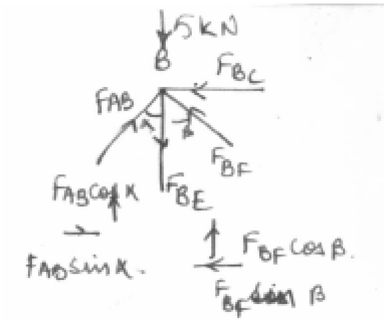
If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (D) as shown in Fig. (b), then shank of the bolt will undergo higher stress. This means that a shank will

	absorb a large portion of the energy, thus relieving the material at the sections near the thread. The bolt, in this way, becomes stronger and lighter and it increase shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us **bolts of uniform strength**. The resilience of a bolt may also be increased by increasing its length.	01 mark 02 marks	04 marks
ii	Types of Shafts: The following two types of shafts are important from the subject point of view: 1. Transmission shafts: These shafts transmit power between the source and machines absorbing power. The counter shafts, line shafts, overhead shafts and all factory shafts are transmission shafts. Since these shafts carry machine parts such as pulleys, gears etc., therefore they subjected to bending in addition to twisting. 2. Machine shafts: These shafts form an integral part of the machine itself. The crank shaft is an example of machine shaft. The material used for shafts should have the following properties: 1. It should have high strength. 2. It should have good machinability. 3. It should have low notch sensitivity factor. 4. It should have good heat treatment properties. 5. It should have high wear resistant properties.	01 mark 01 mark each for any three properti es	04 marks
(b)	The Method of Sections This method is used for the analysis of frames which are: i) symmentrical in nature & ii) Have large no. of members. In the method of sections, a frame is divided into two parts by taking an imaginary "cut" (shown here as x-x) through the frame. Since frame members are subjected to only tensile or compressive forces along their length, the internal forces at the cut member will also be either tensile or compressive with the same magnitude. This result is based on the equilibrium principle and Newton's third law.  Steps for Analysis 1. Decide how you need to "cut" the frame. This is based on: a) where you need to determine forces, and, b) where the total number of unknowns does not exceed three (in general). 2. Decide which side of the cut frame will be easier to work with(minimize	01 mark 01 mark 01 mark for figure 01 mark	

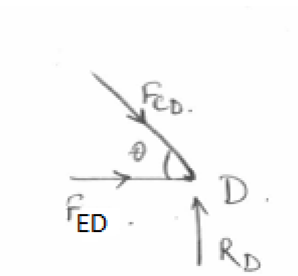
	the number of forces you have to find). 3. If required, determine the necessary support reactions by drawing the FBD of the entire frame and applying the E-of-E. 4.Draw the FBD of the selected part of the cut truss. You need to indicate the unknown forces at the cut members. Initially we assume all the members are in tension, as we did when using the method of joints. Upon solving, if the answer is positive, the member is in tension as per your assumption. If the answer is negative, the member must be in compression. (Please note that you can also assume forces to be either in tension or compression by inspection as was done in the figures above.) 5. Apply the E-of-E to the selected cut section of the truss to solve for the unknown member forces. Note that in most cases it is possible to write one equation to solve for one unknown directly.	01 mark 01 mark 01 mark 01 mark	08 marks
(c)	 Method I:Method of section  Taking moment at point D RA x 5 – 5 x 4 – 5 x 1=0 RA x 5 = 20 +5 RA = 25/5 RA= 5KN But RA + RD =10 RD =5KN Let us consider the equilibrium of truss to lift of section X- X Taking moment at point B Consider FBC,FBE as compressive FFE as tensile force	01 mark for figure 01 mark	

	RA x 1 –FFE x 2 =0 RA = FFE x2 5/2 =FFE FFE = 2.5KN(Tension) Taking moment at point E -RD x + FBC x 2 =0 RD= FBC x 2 FBC=2.5KN(Compression) Method II : Method of joints  Taking Reactions RA + RD =10 Taking Moment at A RD x 5 = 5 x 1 + 5 x 4 RD=5KN RA=5KN Consider joint A  Ө=tan^{-1}(2/1) Σ Fx = 0 FAF–FABcosӨ = 0 FAF = FABcosӨ FAF=2.5KN(Tensile) Σ Fy=0 RA =FABsinӨ FAB= RA / sinӨ FAB= 5.6KN(Comressive) Consider Joint E  Σ Fy= 0 FBF=0.1KN(Tensile) Fx= 0		

-FAE + FFE =0
FFE = FAE
FFE=2.5KN
At Joint B

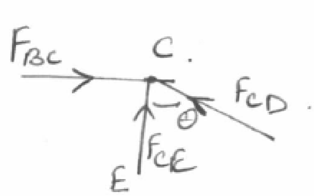


Σ Fx=0
-FBC + FAB sinα- FBFsinβ=0
-FBC + 5.6 x 0.44 – FBF x 0.83=0
FBC + FBF x 0.83=2.464
Σ Fy= 0
-FBE+FAB.cosα + FBF.cosβ-5=0
5.6 x 0.89 + FBF x 0.55=5
FBF=0(App.)
FBC=2.5(Compressive)
At Joint D



½ mark for the calculation of force in each member by any one method i. e. 4½ marks for the forces in 9 members

Σ Fx=0
FED +FCDcosθ=0
FED =-FCDcos θ
Σ Fy= 0
RD=FCDsin θ
FCD=5.6KN(Compressive)
FED=-2.5KN
FED=2.5KN(Tensile)
At Point C



	Σ Fy= 0 FCE+FCD.cosθ =0 FCE= -FCD.cosθ FCE= -5KN FCE= 5KN(Tensile) Force table		
	(see table below)	1½ mark for table	08 marks

Sr.NO.	Member	Force in KN	Nature
1.	AB	5.6	Compressive
2.	BC	2.5	Compressive
3.	CD	5.6	Compressive
4.	DE	2.5	Tensile
5.	AE	2.5	Tensile
6.	BE	0	-
7.	FE	2.5	Tensile
8.	CE	5	Tensile
9.	AF	2.5	Tensile